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Monemian, Emad and Cairns, Alasdair and Pedrozo, V. (2017) Hydrogen and oxygen fuel enrichment effects on a HD diesel engine. Journal of Engine Research, 47 (47). pp. 13-20.

The Journal of Engine Research, Vol. 47 (Summer 2017), pp. 13-20

The Journal of Engine Research

Journal Homepage: www.engineresearch.ir

Hydrogen and Oxygen Fuel Enrichment Effects on a HD Diesel Engine

E. Monemian[1*], A. Cairns[2], V. Pedrozo[3]

[1] *Faculty of Engineering, University of Nottingham, Nottingham, England,* emad.monemian@nottingham.ac.uk
[2] *Faculty of Engineering, University of Nottingham, Nottingham, England,* alasdair.cairns@nottingham.ac.uk
[3] *Department of Mechanical Engineering, Brunel University London, London, England,* vinicius.pedrozo@brunel.ac.uk
**Corresponding Author*

ARTICLE INFO

Article history:
Received: 22 December 2017
Accepted: 21 February 2018

Keywords:
Hydrogen
Oxygen
HD diesel engine
CO_2

ABSTRACT

Implementing emission regulations in the transport sector is enforcing manufacturers to improve performance of Heavy Duty (HD) vehicles as they are accountable for 25% of CO2 emissions within EU. The currently reported research was involved with evaluating partial replacement of hydrogen with diesel fuel experimentally and intake air enrichment with oxygen in a heavy-duty diesel single cylinder engine. Fumigation of hydrogen was done into the intake system at two particular engine loads (6 and 12 bar IMEPn). Indicated efficiency was increased up to 4.6% at 6 bar IMEPn and 2.4% at 12 bar IMEPn, while reducing CO2 emissions by 58% and 32% at 6 and 12 bar IMEPn respectively. Applying diesel pre-injection was required in order to mitigate the high pressure rise rates known as combustion noise. Furthermore, intake air enrichment with oxygen resulted in faster combustion process. This could curb soot and minimised CO emissions to the detriment of NOx increase.

1) Introduction

As of 2008, it must be ensured that the net UK carbon account for the year 2050 is at least 80% lower than the 1990 baseline [1]. Within UK, while the total CO2 was reduced significantly (approx. 30%) since the baseline year 1990 until 2014, the CO2 emission in transport sector was almost unchanged representing 27.5% of total CO2 in 2014 [2]. In that year, CO2 emission of Heavy Goods Vehicles (HGVs) has experienced a 9% improvement compared with 1990. Despite of the fact that UK is on track to meet the second 'carbon budget' regarding the Climate Change Act 2008, transport sector has not contributed a major reduction on CO2 emission due to increase of motor vehicles sales in recent years [3]. As HGVs are accounted for 15.7% of UK transport sector's CO2 emission, the vehicle manufacturers have been required to additionally focus upon Heavy Duty (HD) vehicles [2].

Despite of higher efficiency, Heavy-Duty (HD) suffers from high NOx and soot emissions in addition to CO2. Various methods for soot reduction from such diesel engines have been investigated by many researchers. Extended ignition delay through Low Temperature Combustion (LTC) modes, allows adequate time for mixing prior to the Start of Combustion (SOC). In addition to mitigating combustion noise, this strategy reduces rich regions in the combustion chamber and soot formation is inhibited [4]. However, higher CO and HC emissions are resulted from incomplete LTC combustion. A low-carbon fuel such as hydrogen can be substituted with diesel fuel partially to tackle this issue. The main advantage of burning hydrogen in IC engines is a lack of carbon content, leading to absence of PM, uHC, CO and CO2 exhaust emissions. Nevertheless, the critical obstacles are hydrogen supply and storage. While, hydrogen is produced industrially from methane steam reformation, alternative method involves electrolysis. This method would be a zero carbon route if the required power is supplied by renewable energy; but it is highly priced. Another path for hydrogen power development is storage regarding its safety measures and physical attributes. Alternatives to large storage tanks may be found in hydrides, the materials which can absorb, store and release large quantities of hydrogen gas [6]. High pressure and cryogenic tanks are other options which strive to improve volumetric capacity, conformability and cost.

Various attempts for applying hydrogen in IC engines has been seen during last decades. In 2013, a hydrogen-gasoline unit was implemented on the Aston Martin Rapide S by Alset [7].

Dual-fuelling a 2.2L Puma Diesel with hydrogen was done by Revolve UK in 2008. The hydrogen injector was mounted on the inlet manifold and pilot injection of diesel was used as the ignition source in a permanent dual fuel mode [8].

In 2007, BMW series 7 powered by a hydrogen ICE, could obtain the same efficiency of a baseline turbo-diesel engine [9]. Elsewhere, H2ICE program was followed by Ford since 1997. Ford developed a 2L H2ICE based on Zetec integrated into a P2000 passenger sedan. Consequently, CO2 was decreased to 0.4% that of the gasoline case, with fuel economy improved by 18% [10].

According to Table 1, the significant physical attributes of hydrogen distinguish it from common fuels. Because of too low volumetric energy density at ambient conditions or even in a compressed storage tank, a large volume is required for storing enough hydrogen to give a vehicle adequate driving range.

Table 1: Physical attributes of diesel versus hydrogen [11]

Parameter	Diesel	Hydrogen
Carbon Content (Mass %)	86	0
Stoichiometric air/fuel ratio	14.5	34.3
Ignition Limits [Vol%, λ]	0.6-5.5, 0.5-1.3	4-75%, 0.2-10
Min Ignition Energy at air (λ=1) [mJ]	0.24	0.02
Auto-ignition Temperature [°C]	~250	585
Laminar Flame Velocity at λ=1 [m/s]	0.4-0.8	2.0
LHV [MJ/kg]	42.5	120
Mixture Calorific value at λ=1 [MJ/m^3]	3.83	3.2
Density at 0 °C [kg/m^3]	830	0.089

Hydrogen has a wide flammability range (4-75% vol. concentration in air) versus all other fuels, which enables combustion in an IC engine over a wide range of fuel-air mixtures. Lower flammability limit (LFL) plays as a turning point in dual-fuel combustion as the mechanism is affected depending on which side of LFL, the hydrogen concentration is. To clarify this point, a conceptual model proposed within a relevant work by Morgan et al. [5] including the following three modes was considered (Figure 1):

1. When hydrogen concentration is above its LFL, hydrogen is pre-ignited resulting in an auto-ignition like HCCI mode or knocking combustion type.
2. In case of lean hydrogen (below LFL), hydrogen just burns in existence of diesel diffusion flame in a mixing-controlled mode.
3. If hydrogen concentration is over the LFL and in-cylinder conditions are not providing the hydrogen burn prior to diesel fuel ignition, the premixed hydrogen - air combustion develops in laminar mode encircling the diesel diffusion flame.

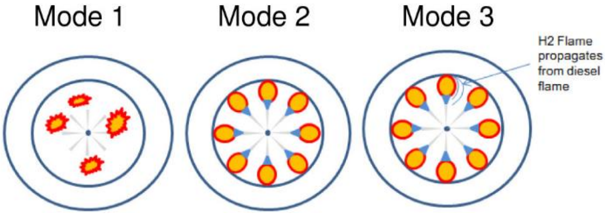

Figure 1: Hydrogen-diesel conceptual combustion model [5]

In addition, high flame velocity of hydrogen results in more efficient operation versus gasoline. Due to very high diffusivity of hydrogen, forming a uniform mixture of fuel and air is easily facilitated. This is also advantageous in the case of a hydrogen gas leakage, with rapid dispersion. Due to a high temperature of auto-ignition, hydrogen cannot be used directly in a CI engine. Thus, an amount of diesel fuel is needed to ignite the gas via "pilot ignition". Hydrogen's low ignition energy and high burning speed makes a hydrogen/diesel mixture easier to ignite, reducing misfire and thereby improving performance and emissions. In terms of power output, hydrogen enhances the mixture's energy density at lean mixtures by increasing the H/C ratio [12].

Since hydrogen has almost three times higher specific energy by mass (LHV) compared to diesel, a substantial part of demanded diesel fuel can be substituted by hydrogen. However, diverse challenges remain including high in-cylinder pressure rise rates and the occurrence of pre-ignition. Flashback conditions can also develop if the pre-ignition occurs near the intake valve and the resultant flame travels back into the intake system, particularly under heavy loads. Hydrogen's high flame velocity is aids in terms of knock. Nevertheless, flashback can be caused by in-cylinder hotspots within intake stroke as hydrogen has very low ignition energy [12].

The current paper has been involved with evaluating experimental hydrogen and oxygen enrichment of a heavy-duty diesel engine.

2)Experimental Methodology

The test engine used for the experiments was an externally boosted single cylinder HD diesel engine which resembles the engine of a typical current UK heavy goods vehicle (HGV). The specifications of the test engine is detailed in Table 2.

Intake air pressure was boosted via an external compressor while limited control over the intake air flow was provided by an intake throttle after the intake air mass flow meter. Cooled EGR mixed with the boosted intake air inside the intake surge tank.

The mass flow controller regulated the hydrogen fuel being supplied from a gas cylinder. With aim of protection against hydrogen leakage, gas detection and emergency shutdown system were embedded inside the test cell. In next stage, oxygen which was used for intake air enrichment by up to 20%, was regulated to 3.5bar and its flow was controlled by a rotameter (RM&C®). Also separate flashback arrestors were fitted to both hydrogen and oxygen supply. In addition to the schematic of experimental setup (Figure 2), the hydrogen and oxygen fumigation system is displayed in Figure 3.

Table 2: Test Engine Specifications

Parameter	Value
Bore × Stroke	129 mm × 155 mm
Connecting rod length	256 mm
Swept volume	2.026 dm^3
Number of valves	4
Compression ratio	16.8 : 1
Max in-cylinder pressure	180 bar
Diesel injection system	Bosch common rail, 220 MPa max injection pressure, 8 holes, 150° spray
Diesel fuel	Diesel – off-road "red" diesel (LHV = 42.9 MJ/kg)
Hydrogen / Oxygen Enrichment	Continuous fumigation into intake port
Hydrogen material	BOC® CP grade hydrogen N5.0 (LHV = 120 MJ/kg)
Oxygen material	BOC® Oxygen N2.6

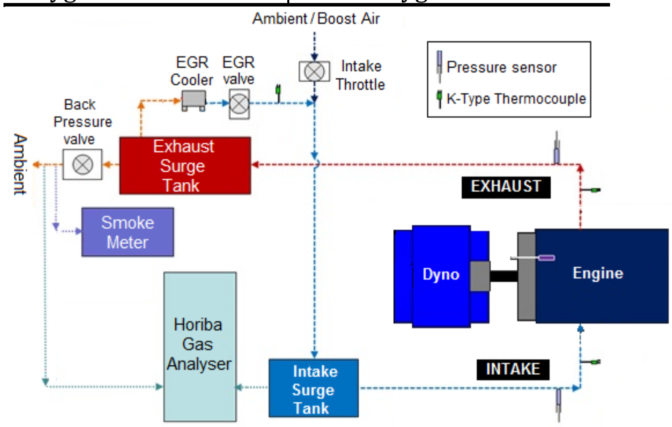

Figure 2: Schematic of experimental setup [15]

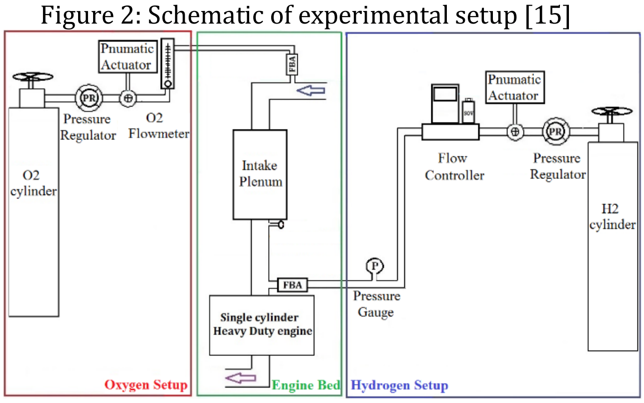

Figure 3: Hydrogen and oxygen fumigation setup

The hydrogen substitution ratio (HF) was calculated based on input energy as seen in Eq. (1):

$$HF = \frac{\dot{m}_{hydrogen} LHV_{hydrogen}}{\dot{m}_{hydrogen} LHV_{hydrogen} + \dot{m}_{diesel} LHV_{diesel}} \quad (1)$$

The global equivalence ratio (H2+diesel/air mixture) is calculated by the Eq. (2):

$$\emptyset_{global} = \frac{stoich\ AFR_{H2}\dot{m}_{H2}+stoichAFR_{diesel}\dot{m}_{diesel}}{\dot{m}_{air}} \qquad (2)$$

DAQ (data acquisition) system was programed in-house for monitoring and recording the data acquired by two National Instrument cards. Some of the instrumentation data which are synchronised with an encoder (resolution = 0.25 crank angle degrees), are acquired by the high speed DAQ card. While the remained low frequency operating signals were obtained by the low speed DAQ card. Heat release analysis was based on the first law of thermodynamics and ideal gas law (Heywood Eq. 9.26 [14]). While the heat losses to crevices and cylinder walls were combined with the chemical heat release, the combination is termed as the "net heat release" and can be calculated as follows:

$$\frac{dQ_n}{dt} = \frac{\gamma}{\gamma-1}p\frac{dV}{dt} + \frac{1}{\gamma-1}V\frac{dp}{dt} \qquad (3)$$

where the ratio of specific heats was assumed constant of γ = 1.33. Cyclic variability was defined by the coefficient of variation of the net IMEP (COV_IMEPn), averaged over 200 sampled cycles. While a Kistler® piezoelectric pressure transducer sensed the in-cylinder pressure, intake and exhaust pressures were measured by piezo-resistive pressure sensors. Pressure sensors and thermocouples locations are laid out in Figure 2.

Engine-out gaseous emissions (NOx, HC, CO and CO2) measurements were taken using a Horiba MEXA 7170DEGR. AVL 415SE smoke meter was used for soot emission sampling. The peak average pressure rise rate (PRR) and COV_IMEPn limits were set to 20 bar/CAD and 5%, respectively.

3) Test Conditions

Test work was done in two stages: 1) substituting diesel fuel with hydrogen 2) enriching intake air with oxygen. In first testing stage, two specific operating points were chosen. The first operating point was equivalent to 25% load (6 bar IMEPn) and the second operating point was equivalent to 50% load (12 bar IMEPn). These test points are representative of the A25 and A50 of European Stationary Cycle (ESC13) as seen in Figure 4.

Hydrogen substitution ratio in each operating point, was limited by maximum flow rate of mass flow controller (100 lit/min).

In second testing stage, hydrogen substitution ratio was fixed at HF=20%. Pure oxygen was fumigated into the intake air pipeline in order to vary the actual O2 concentration of the intake charge. The O2 enrichment levels were equivalent to ~10% and 20% of the initial O2 concentration.

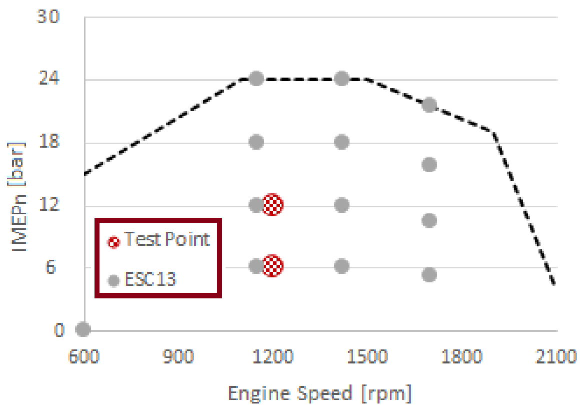

Figure 4: ESC13 cycle points and test points superimposed on the HD diesel engine operational map [15]

Table 3: Testing Stage 1 Operating Conditions: H2 Substitution

Parameter	Test Point 1	Test Point 2
Speed	1200 rpm	1200 rpm
IMEPn	6 bar	12 bar
H2 Energy Fraction Range	0% to 65%	0% to 35%
EGR Rate	25%	25%
EGR Temperature	339 K	367 K
Intake Air Temperature	309 K	318 K
Intake Pressure	1.25 bar	1.90 bar
Exhaust Pressure	1.35 bar	2.00 bar
Rail Pressure	1250 bar	1400 bar

Table 4: Testing Stage 2 Operating Conditions: O2 enrichment of intake air

Parameter	Test Point 1	Test Point 2
Speed	1200 rpm	1200 rpm
IMEPn	12 bar	12 bar
H2 Energy Fraction	20%	20%
EGR Rate	0%	30%
EGR Temperature	-	372 K
Intake Air Temperature	320 K	320 K
Intake Pressure	1.90 bar	1.90 bar
Exhaust Pressure	2.00 bar	2.00 bar
Rail Pressure	1400 bar	1400 bar

4)Results and discussion

Stage 1: H2 substitution

Firstly, it is worth to mention about the diesel injection strategy followed during H2 fumigation. As seen in Figure 5, very high in-cylinder pressure rise was resulted from the single injection at both A25 and A50. In contrast, pre-injection of diesel pilot smoothed HRR and lowered pressure rise rate (PRR) due to advancing SOC and reducing ignition delay. Therefore, pre-injection was chosen as a strategy to reduce the combustion noise for all test cases while the optimised start of injection (SOI) was kept constant for both pilot and main diesel injection.

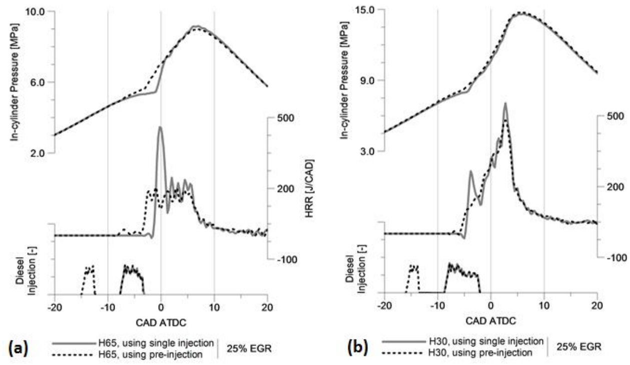

Figure 5: Combustion noise reduction using pre-injection (a) A25 (b) A50

Initially, effect of H2 substitution on combustion process was studied. According to Figure 6, while CA10-CA50 and CA50 are fairly resistant to H2 fraction at A25, combustion timing and first half burning was slightly accelerated for higher HFs at A50. This relates to higher load condition and hydrogen stimulant attribute more pronounced at A50 rather than A25. Nevertheless, overall combustion duration is longer in 12 bar IMEPn, regarding CA10-CA90 plot.

Although, combustion duration for both loads shows a parabolic trend with a peak around HF = 15% for A50 and HF = 25% for A25, but it was shortened by substituting more hydrogen as H2 accelerated the combustion. This is due to higher flame speed of hydrogen which is reflected in shorter CA10-CA50 duration at A50.

The global equivalence ratio calculated by Eq. (2), shows lower (almost constant) value for A25 mainly due to lower diesel fuel to air ratio cf. A50. Volumetric efficiency and exhaust gas density dropped as more hydrogen was fumigated, as it replaces intake fresh air and presents a lower molecular weight combined with a higher LHV. On the other hand, although both loads showed acceptable (fairly constant) COV_IMEP, a lower cyclic variation was observed at A50 versus A25. This might be due to less complete hydrogen combustion at lower load.

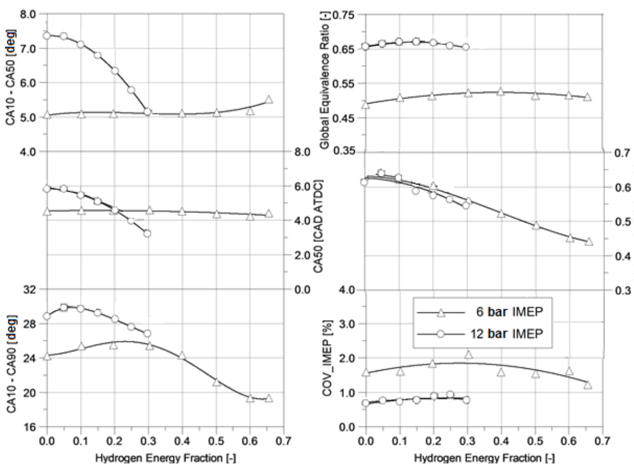

Figure 6: In-cylinder conditions (testing stage 1)

Considering H2-diesel combustion emissions, propitious effect of hydrogen fumigation on CO2 and most carbon-contained emissions was seen, as targeted more specifically in this wok, Figure 7. By increasing hydrogen fraction, while ISHC remained fairly constant, ISCO2, ISCO and ISSoot all decreased drastically at both loads. The main reason relates to lower carbon to hydrogen ratio content of the dual-fuel. However, ISNOx was increased with higher mass flow of hydrogen which was expected due to combustion accelerating of hydrogen leading to higher cylinder temperatures.

According to Figure 7, hydrogen enrichment in lower volumes resulted in slight drop in indicated efficiency, possibly joint with reduction of specific heats ratio because of air displacement. Though, quicker combustion overshadowed this influence at higher HFs resulting in the efficiency amelioration. Indicated efficiency had an unfavourable effect in lower hydrogen volumes till HF = 30% at A25 and HF = 10% at A50. This is due to incomplete hydrogen combustion (hydrogen slip) based on the work reported in [5]. Thereafter, indicated efficiency starts to rise monotonically until it reaches the peak values at the highest attainable HF for both loads.

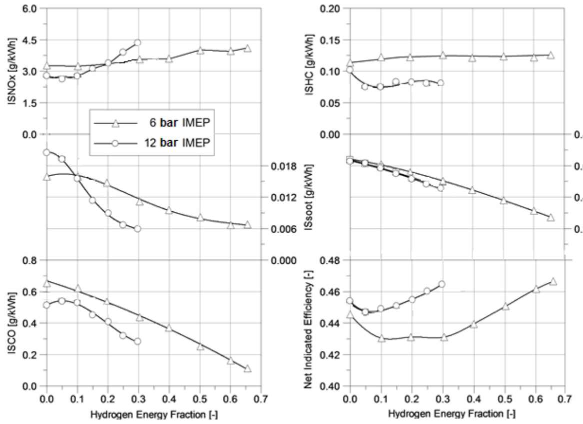

Figure 7: Emissions and Indicated Efficiency (testing stage 1)

Stage 2: intake air enrichment with O2

In second testing stage, effects of enriching the intake air by 10% and 20% of initial O2 concentration was observed. Figure 8 shows effect of O2 enrichment on in-cylinder pressure and HRR. As expected, EGR had mitigated the maximum pressure and rate of heat release in comparison to non-EGR mode. Besides, O2 addition caused slight increase to these parameters due to providing a leaner combustion as global equivalence ratio shows in Figure 9.

As seen in Figure 9, O2 enrichment has shortened slightly the combustion duration due to leaner in-cylinder condition. In addition, combustion phasing and CA10-CA50 duration were decreasing. This trend is well presented with EGR rather than without it. It is obvious that dilution effect of EGR caused overall deceleration in burn process.

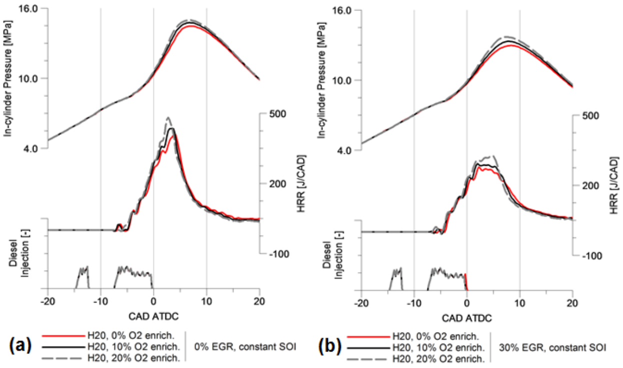

Figure 8: O2 enrichment effects: (a) 0% EGR (b) 30% EGR (testing stage 2)

On other hand, EGR mitigated the pressure rise and stabilised the combustion in comparison to the EGR mode. In addition, EGR effect was bolded in restraining the ISNOx as seen Figure 10. In non-EGR mode, while all emissions except ISNOx were reluctant to O2 enrichment, ISCO and ISsoot were reduced significantly in EGR mode. Ultimately, although indicated efficiency was increasing with O2 enrichment in EGR mode, it was decreasing in non-EGR mode.

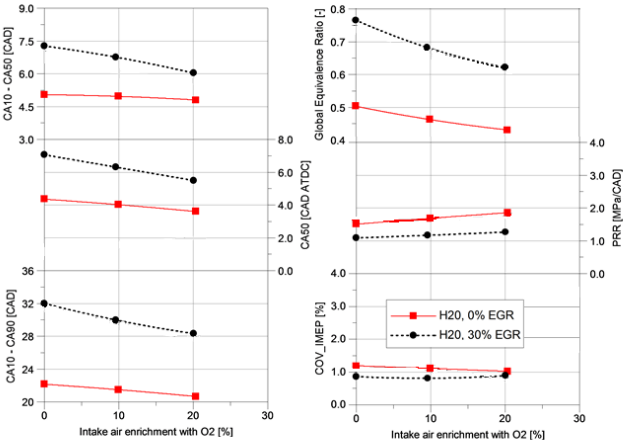

Figure 9: In-cylinder conditions (testing stage 2)

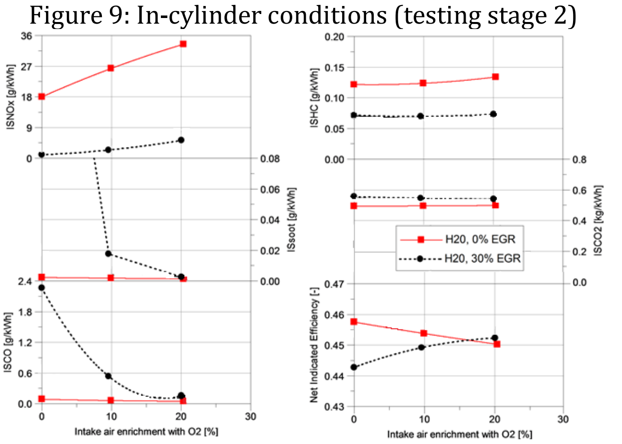

Figure 10: Emissions and indicated efficiency (testing stage 2)

Conclusions

The currently reported work was involved with experimental evaluation of H2 and O2 enrichment in a HD diesel engine in respects of combustion, efficiency and emissions. The experiments were taken in two testing stages: 1) substituting diesel fuel with hydrogen 2) enriching intake air with oxygen. The following conclusions were made:

- The pre-injection prior to the main diesel injection reduced the levels of PRRs during hydrogen fumigation.
- In stage 1, the highest obtained hydrogen fractions made indicated efficiency to increase up to 4.6% at 6 bar IMEPn (HF = 65%) and 2.4% at 12 bar IMEPn (HF = 35%), respectively.
- In stage 1, CO2 emissions were reduced by 58% and 32% at 6 and 12 bar IMEP respectively, using highest HF. Soot and CO emissions were reduced as more hydrogen was injected, particularly at the lower load of 6 bar IMEP.
- In stage 2, the O2 enrichment strategy curbed smoke and minimised CO emissions to the detriment of considerable NOx increase.
- As expected, EGR dilution effect had decelerated the overall burn process within stage 2. Also, ISNOx was restrained by applying EGR.

Acknowledgement

The experimental testing was sponsored by VN Automotive Ltd.

References

[1] Climate Change Act 2008: http://www.legislation.gov.uk/ukpga/2008/27/pdfs/ukpga_20080027_en.pdf

[2] Final UK greenhouse gas emissions national statistics 1990-2014: https://www.gov.uk/government/statistics/final-uk-greenhouse-gas-emissions-national-statistics-1990-2014

[3] http://www.oica.net/category/sales-statistics

[4] R.D. Reitz, G. Duraisamy, Review of high efficiency and clean reactivity controlled compression ignition (RCCI) combustion in internal combustion engines, Progress in Energy and Combustion Science, 46 (2015) 12-71

[5] R.E. Morgan, P. Atkins, A. Atkins, C. Lenartowicz, M. Heikal, Effect of Hydrogen Fumigation in a Dual Fueled Heavy Duty Engine, SAE paper 2015-24-2457

[6] NWO (Netherlands Organization for Scientific Research). Light weight hydrogen 'tank' could fuel hydrogen economy. ScienceDaily, www.sciencedaily.com/releases/2008/11/081104084215.htm , accessed 5 November 2008

[7] Alset Global "Our Solutions", retrieved 18 May 2017 http://alset.at/our-solutions/

[8] J. Kendall, Hydrogen ICE conversion targets cargo vans, Automotive Engineering article, SAE Intl. edited: 19 Dec 2008

[9] http://en.wikipedia.org/wiki/BMW_Hydrogen_7

[10] S. Szwabowski, S. Hashemi, W. Stockhausen, R. Natkin, L. Reams, D. Kabat, C. Potts, Ford Hydrogen

Engine Powered P2000 Vehicle, SAE paper 2002-01-0243, 2002
[11] Periodic Table of Elements-Hydrogen: http://environmentalchemistry.com/yogi/periodic/H.html
[12] Hydrogen Use in Internal Combustion Engines, College of the Desert, Dec 2001
[13] V.B. Pedrozo, I. May, M. Dalla Nora, A. Cairns, H. Zhao, Experimental analysis of ethanol dual-fuel combustion in a heavy-duty diesel engine: an optimisation at low load, Applied Energy, 165 (2016): 166–182.
[14] J.B. Heywood, Internal combustion engine fundamentals, 1st edition, McGraw-Hill, 1988
[15] E. Monemian, A. Cairns, M. Gilmore, D. Newman, K. Scotts, Evaluation of intake charge hydrogen enrichment in a heavy-duty diesel engine, Proceeding of IMechE Part D: Journal of Automobile Engineering, Nov 2017, DOI: 10.1177/0954407017738375

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اثر غنی سازی سوخت موتور دیزل با هیدروژن و اکسیژن

عماد منعمیان[۱*]، السدر کرنز[۲]، وینیسیوس پدروزو[۳]

[۱] دانشکده مهندسی، دانشگاه ناتینگهام، ناتینگهام، انگلستان، emad.monemian@nottingham.ac.uk

[۲] دانشکده مهندسی، دانشگاه ناتینگهام، ناتینگهام، انگلستان، alasdair.cairns@nottingham.ac.uk

[۳] گروه مهندسی مکانیک، دانشگاه برونل، لندن، انگلستان، vinicius.pedrozo@brunel.ac.uk

*نویسندۀ مسئول

چکیده

پیاده سازی قوانین آلایندگی در بخش حمل و نقل، سازندگان را ملزم به بهبود عملکرد موتورهای دیزل سنگین کرده است. چرا که این وسایل نقلیه مسبب ۲۵٪ انتشار دی اکسید کربن در اتحادیه اروپا می باشند. تحقیق حاضر به بررسی تجربی جایگزینی بخشی از سوخت دیزل با گاز هیدروژن و غنی سازی اکسیژن هوای ورودی یک موتور دیزل سنگین تک استوانه می پردازد. بخوردهی هیدروژن در هوای ورودی در دو بار کاری ویژه (۶ bar imep و ۱۲ bar imep) صورت پذیرفته است. نسبتهای بالای جانشینی هیدروژن باعث افزایش بازده اندیکاتوری به میزان ۴٫۶٪ و ۲٫۴٪ به ترتیب در بارهای ۶ bar imep و ۱۲ bar imep گردیده است. در حالیکه این امر باعث کاهش دی اکسید کربن خروجی به میزان ۵۸٪ و ۳۲٪ به ترتیب در بارهای ۶ bar imep و ۱۲ bar imep می گردد. پاشش مقطعی قبل از پاشش اصلی سوخت دیزل منجر به کاهش نرخ بالای رشد فشار استوانه موسوم به اختلال احتراق می شود. همچنین، غنی سازی اکسیژن هوای ورودی باعث تسریع احتراق شده، دوده و مونواکسید کربن خروجی را به هزینه افزایش چشمگیر اکسید نیتروژن کاهش می دهد.

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